Residential Central Air-Conditioner, Small Duct, High Velocity (SDHV) Systems Standards Rulemaking

Engineering Analysis

Draft Report for Review

Prepared for:

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<u>Foreword</u>

The U.S. Department of Energy (DOE) issued a final rule on May 23, 2002 (67 FR 36368), increasing the minimum efficiency standards for most residential central air conditioners and heat pumps to 12 SEER/7.4 HSPF. However, during the rulemaking process, DOE received information indicating that the special characteristics of small duct, high velocity (SDHV) air conditioner and heat pump systems make it unlikely that such systems could meet minimum efficiency standards established for conventional products. Consequently, both the SDHV manufacturers and the industry trade association, the Air-Conditioning & Refrigeration Institute (ARI), supported the creation of a separate product class for SDHV systems and the development of technologically feasible and economically justified standards for this product.

In the May 23, 2002, final rule, DOE agreed that SDHV systems should not be subject to the standards set for conventional products. DOE established a separate product class for SDHV systems based on the following definition:

"<u>Small duct, high velocity system</u> means a heating and cooling product that contains a blower and indoor coil combination that: (1) Is designed for, and produces, at least 1.2 inches of external static pressure when operated at the certified air volume rate of 220-350 CFM per rated ton of cooling; and (2) When applied in the field, uses high velocity room outlets generally greater than 1000 fpm which have less than 6.0 square inches of free area."

In the May 23, 2002 final rule, DOE retained the 10 SEER / 6.8 HSPF standards established under the National Appliance Energy Conservation Act (NAECA) for SDHV systems, pending further study to establish appropriate higher standard levels. DOE also stated its intention to publish a final rule for the test procedure in which minimum efficiency standards for SDHV systems will be based on the testing requirements developed for SDHV systems in the air conditioning test procedure revision currently being finalized, or in a future revision specifically aimed at SDHV products.

DOE determined that additional analysis on the cost and technical issues related to SDHV air conditioner and heat pump products was needed to determine appropriate minimum efficiency standards for this class of product. DOE's plan for establishing the manufacturing cost and efficiency relationship for SDHV systems was to involve the manufacturers that produce these products.

The intention of this report is to document the methodology and assumptions used in the Engineering Analysis which develops the relationship between cost and efficiency for SDHV systems. A separate spreadsheet[1] enables a user to reproduce the results of the Engineering Analysis using the current assumptions or to revise the assumptions. The cost-efficiency relationship from the Engineering Analysis is an input to the *Life-Cycle Cost (LCC) Analysis* discussed in a separate report. A spreadsheet for the LCC Analysis[1] also allows a user to reproduce the LCC results.

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[1] http://www.eren.doe.gov/buildings/codes_standards/applbrf/central_air_conditioner.html

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1. INTRODUCTION

1.1 SDHV Systems Description

Small-duct, high-velocity (SDHV) systems are a type of air-cooled, split-system central air conditioning and air conditioning/heat pump systems considered by DOE to be "space-constrained," thus requiring special consideration in determining the technical feasibility and economic justification criteria required for a new efficiency standard.

SDHV air conditioning systems currently marketed in the U.S. are similar to conventional split system air conditioners that consist of an outdoor condensing unit and an indoor coil unit, with or without an integral blower. A unique feature of SDHV systems is the air distribution system. The air distribution system together with the blower-coil unit is usually installed in attic spaces of older homes that were not designed for forced-air circulation heating or cooling. Such homes lack conventional (i.e., large-size, low velocity) ducts, which are costly and difficult to retrofit. Thus, installation of a SDHV system helps satisfy the air conditioning needs for these older homes, replacing less efficient window air conditioners.

The air distribution system is comprised of a ceiling-mounted return grill and return air duct which delivers room air to the blower-coil unit and a supply air system which delivers conditioned air from the blower-coil unit to various spaces within the building. The supply system consists of an insulated, rigid plenum, 7 to 9 inches in diameter, usually running the length of the attic space. A series of flexible two-inch diameter branch ducts, which are small enough to be concealed within wall spaces, are connected to the plenum. Each branch duct terminates with a supply air outlet. The ceiling-mounted outlets deliver supply air at a velocity of at least 1000 feet per minute (fpm) where the high velocity air enters, entraining and mixing with room air.

The outdoor condensing units used in SDHV systems are typically sourced from a full line manufacturer of conventional split systems. These condensing units are identical to those used by the full-line manufacturers in their product lines of conventional systems. A SDHV heat pump system is comprised of a conventional outdoor heat pump unit and an indoor blower-coil unit and duct system similar to that used for cooling, except that the blower-coil unit is equipped with a larger coil for refrigerant management and a back-up heat source, such as a resistence heating element or hot water coil, is added.

1.2 Comparison with Conventional Split Systems

The blower-coils used in SDHV systems are different from those used in conventional split systems. The significant differences are discussed in this section.

1.2.1 Coil Geometry

The indoor coils of SDHV systems, while constructed of copper tubing and aluminum fins and having design features similar to that of indoor coils for conventional systems, usually have a smaller face area and contain more rows of coil. A key performance attribute for a SDHV system is called its “core volume,” which is the product of the number of coil rows and the coil face area. Since the number of coil rows is a surrogate for coil depth, the product of coil rows and face area is related to volume (and weight). As will be discussed in Sections 2.4.1 and 2.6.1, core volume has a major impact both on performance and cost. Indeed, the relationship between core volume of SDHV systems and core volume of matched indoor coils, which can vary over a wide range, has a major impact on SDHV system performance.

1.2.2 Air Flow Rate

Conventional air conditioning systems are usually designed to operate with an indoor air volumetric air flow rate per rated ton (12,000 Btu/h) of cooling capacity at 400 cubic feet per minute (CFM) per ton. In contrast, SDHV systems are designed to operate at ranges from 50 to 70 percent of the nominal 400 CFM/ton of conventional systems, or 200 to 280 CFM per nominal ton. However, matching an SDHV blower coil unit with a conventional outdoor unit reduces its rated capacity by 10 to 20 percent. Therefore, the CFM per actual ton of a SDHV system is in the range of 220 to 310 CFM/ton.

1.2.3 Blower Static Pressure

Because of the small duct size of SDHV systems, air velocity is much greater than with conventional systems, resulting in higher frictional loss in the air distribution system. SDHV systems are rated for performance with a minimum external static pressure of 1.20 inches of water. Conventional systems are rated for performance with a minimum external static pressure that ranges from 0.10 to 0.20 inches of water, depending on system capacity.

The air-side pressure drop across the SDHV indoor coil is approximately equal to that of conventional systems, which for a wet coil is typically about 0.30 inches of water and for a dry coil is typically about 0.20 inches of water.

1.2.4 System Power Consumption

The blower coil unit for SDHV systems operates with reduced air flow compared to conventional air conditioning systems and supplies chilled air to the occupied space through a system of small-size ducts. Because of the reduced volumetric rate of air flow through the blower coil units' evaporator, air temperature delivered to the conditioned spaces is lower (and its humidity ratio is lower) than that of conventional systems. The reduced air flow through the evaporator also decreases the refrigerant pressure leaving the evaporator and returning to the compressor in the outdoor unit, which decreases the compressor's capacity and efficiency. Also, the higher friction in the small ducts increases the work performed by the blower, which increases the blower motor's power consumption. Because of the higher compressor and blower power, a SDHV system usually operates with a lower efficiency and capacity than those of the matched system. Thus, to achieve the current 10 SEER minimum efficiency, a SDHV blower-coil must often be matched with a condensing unit which is rated at higher efficiency and capacity. Consequently, SDHV systems are less amenable than conventional systems to increases in efficiency standards.

1.3 Industry

Two companies produce SDHV systems: Unico, Inc. and the SpacePak Division of Mestek Incorporated. These companies typically out-source production of all components of the blower coil units and function as assemblers. They also out-source production of most components used in the duct distribution system, manufacturing only the components that are unique to their designs. More information is available on the manufacturer web sites at http://www.UnicoSystem.com and http://www.SpacePak.com.

The total market for SDHV systems is estimated to be less than 20,000 units/year[2], or about one-half of one percent of the total market for residential central air conditioning systems. Because SDHV blower-coil components are currently outsourced, the impacts of increased efficiency standards on the production facilities of SDHV manufacturers is likely to be small[3].

[2] Estimate based on private communications with manufacturers.

[3] Determination of impacts of standards on SDHV manufacturers is not within the scope of this analysis.

2. ENGINEERING ANALYSIS

The Engineering Analysis develops the relationship between the efficiency and cost of a central air conditioner or heat pump. This relationship serves as the basis for the subsequent life-cycle cost analysis. Determining the cost-efficiency relationship involves analysis of the options available to manufacturers for increasing the efficiency of a baseline product (i.e., one that just meets the minimum efficiency standard). In general, the options available to SDHV manufacturers which result in higher SDHV system efficiencies are limited to a) design changes or operational constraints to blower coil units; b) pairing of existing SDHV blower coil models with more efficient outdoor units; and c) a combination of options a and b. Because of the small market share of SDHV blower coil units, unitary systems manufacturers do not have strong incentives to redesign their outdoor units for optimal SDHV applications, therefore, outdoor unit redesign is not considered a viable option.

2.1 Data Sources

The Engineering Analysis for SDHV systems used technical and cost data obtained from both public and private sources. The public data used included published materials from Chapter 4 of the Technical Support Document (TSD) produced for the Central Air Conditioning (CAC) rulemaking[4], data from the ARI certification data base, and materials published in manufacturer catalogs. The private data included non-published information obtained by Arthur D. Little, Inc., obtained under non-disclosure agreements with several air conditioner manufacturers during the CAC rulemaking and data obtained by Navigant Consulting, Inc., under non-disclosure agreements with Unico Inc. and the Spacepak division of Mestek, the two SDHV systems manufacturers.

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[4] http://www.eren.doe.gov/buildings/codes_standards/applbrf/central_air_conditioner.html

All the assumptions, results and formulas used in the Engineering Analysis, with the exception of data covered by non-disclosure agreement, are available for stakeholders' review in the form of a worksheet posted on the DOE web site.[5] The worksheet is designed such that users can verify assumptions, analyze sensitivities, review methodologies and submit comments. Additional information is contained in the worksheet itself.

2.2 Modeling SDHV System Performance

System performance, along with cost estimates, is one of the two key factors in the Engineering Analysis. To estimate SDHV system performance, DOE took advantage of an existing rating method that is currently available for use by industry to rate split-system air conditioners. The rating method is particularly useful for the intended purpose because its use avoids the need for testing, computer simulation or equipment teardown.

In rating split-system air conditioners and heat pump systems for efficiency, DOE requires a unitary system manufacturer to test the highest likely sales volume combination of indoor and outdoor units they produce[6]. For other indoor-outdoor unit combinations produced by the same manufacturer, or in part by a component manufacturer using the same outdoor unit, DOE allows the efficiency rating to be determined by an alternative rating method. The alternative rating method, which must be requested by a manufacturer and approved by DOE, must be supported by test data, as specified in the Code of Federal Regulations. One such rating method, developed by the National Institute of Standards and Technology (NIST), is accepted by DOE.

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[5] http://www.eren.doe.gov/buildings/codes_standards/applbrf/central_air_conditioner.html

[6] 10 CFR part 430, section 430.24(m)(2)

2.2.1 NIST Rating Method for Split Air Conditioning Systems

The NIST rating method for air conditioning systems is described in the U.S. Department of Commerce Report, NISTIR 89-4071[7]. The NIST rating method uses measured performance data for a "matched system" or tested combination of outdoor and indoor units and calculates the performance with a different indoor unit substituted for the tested indoor unit. The non-tested combination is called a "mixed system." The technical basis for the NIST rating method is described elsewhere.[8] The stated purpose of the NIST rating method is to account for the interaction of three components deemed most likely to be changed in a matched system: the indoor evaporator coil, the indoor fan, and the expansion device. Specific equations and constraints for the NIST rating method are excerpted from the 1989 NIST report and provided in Appendix A.

2.2.2 NIST Rating Method Specialized for SDHV Rulemaking

This section describes the specific assumptions made in the Engineering Analysis for the purpose of estimating the performance (capacity and efficiency) of a mixed system consisting of a SDHV blower-coil combined with a conventional condensing unit. These assumptions specialize the general equations of the NIST method and are coded as equations and data in an accompanying spreadsheet. Because of the restrictive nature of these assumptions and the specificity of the data, use of the spreadsheet is not recommended for purposes other than that of the rulemaking.

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[7] Domanski, Piotr A., "Rating Procedure for Mixed Air-Source Unitary Air Conditioners and Heat Pumps in the Cooling Mode-Revision 1, May 1989.

[8] Domanski, Piotr A., "Rating of Mixed Split Air Conditioners", Proceedings of 5th Annual Symposium on Improving Energy Efficiency in Hot and Humid Climates, Houston, TX; September, 1988.

2.2.2.1 System Assumptions Specific for SDHV Rulemaking Engineering Analysis

Matched system:

1. Air flow rate	400 CFM/nominal ton
2. External static pressure	0.15 in. H_2O
3. Coil air side pressure drop	0.3 in. H_2O
4. Blower/motor efficiency	45 percent (Default)
5. Indoor fan power	0.365 W/CFM

SDHV (mixed) system:

6. Air flow rate	220-350 CFM/rated ton
7. External static pressure	1.2 in.H_2O
8. Coil air side pressure drop	0.3 in. H_20
9. Blower/motor efficiency	65 percent

Matched and mixed systems:

10. Both equipped with similar type thermostatic expansion valves,
11. Both have similar part load efficiency factors, C_D,
12. Both indoor coils use similar coil technology, e.g., same tube materials, geometries, enhancements; and same fin materials, geometries, and enhancements.

2.2.2.2 Other Assumptions

The distinguishing feature of the SDHV methodology is the manner in which values for the indoor fan power, $P_{F,,x}$ and indoor coil scaling factor, $F_c = Q_{c,x}/Q_{c,m}$, are determined for use in the NIST equations (Appendix A). In this section the assumptions are refined and differences with the NIST method are noted. In section 2.2.2.3, the rationale behind these assumptions is discussed.

<u>*SDHV Fan Power, $P_{F,x}$*</u>

Since most SDHV blower coils are paired with matched systems which are rated based on a cased indoor coil (i.e., coil only), the matched system indoor fan power is provided by the default value as:

$$P_{F,m} = 0.365 \times CFM_m, \qquad (2.1)$$

where CFM_m is the nominal volumetric air flow rate of the matched system, given by

$$CFM_m = 400 \times Q_m /12,000. \qquad (2.2)$$

Note that equation 2.2 differs from NIST equation 4.6 (see Appendix A) because 400 CFM/ton is the nominal volumetric air flow in general use by industry when rating coil-only systems where the air flow is supplied by furnaces.

Blower motors convert electrical energy into work done on the air stream. For blower motors used in heating, ventilation and air conditioning (HVAC) applications, work is proportional to the product of the volumetric air flow rate and the fan total pressure rise. The ratio of the work done on the air stream to the electrical energy input to the motor is the blower-motor's "wire-to-air" efficiency, η. For the SDHV system and the matched system, the total pressure of the fan is the sum of the static pressure losses in the coil and in the supply ducts. Thus, the SDHV system fan power is determined from the matched system fan power by:

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$$P_{F,x} = P_{F,m} \, x \, [CFM_x \times (ESP_x + CPD_x)]/[CFM_m \times (ESP_m + CPD_m)] \, x \, \eta_m/\eta_x \quad (2.3)$$

where:

CFM_x is the selected indoor air volumetric flow rate of the mixed (SDHV) system,

ESP_m is the external static pressure imposed on the indoor unit of the matched system prescribed by rating standards (listed in section 1.2.3),

ESP_x is the external static pressure imposed on the indoor unit of the mixed (SDHV) systems,

CPD_m is the air side pressure drop of the wet indoor coil for the matched system,

CPD_x is the air side pressure drop of the wet indoor coil of the mixed system,

η_x is the combined blower and motor efficiency of the SDHV system,

η_m is the combined blower and motor efficiency of the matched system.

Indoor coil scaling factor, $F_c = Q_{c,x}/Q_{c,m}$

The NIST method requires the capacities of the matched and mixed coils be obtained using a "verified method" at the same set of thermodynamic conditions for the air and refrigerant. The air side flow rates are required to be the same as those specified for the mixed and matched systems. Verified methods include manufacturers' test data or computer simulations based on fundamental principles of heat transfer phenomenon.

For the purpose of the SDHV Engineering Analysis, simplifying assumptions are essential because meeting the exact specifications of the NIST method for verification is not considered practical. Based on "rules of thumb" frequently used by design engineers, DOE assumes that the indoor coil scaling factor is given by:

$$F_c = [(CFM_x/CFM_m) \times (FA \times Rows)_x/(FA \times Rows)_m]^{0.5} \qquad (2.4)$$

Where:

$(FA \times Rows)_x / (FA \times Rows)_m$ is the ratio of "core volumes" (product of face area and number of rows) for the SDHV coil to that of the matched system indoor coil.

2.2.2.3 Discussion of Assumptions

SDHV Fan Power, $P_{F,x}$

As previously discussed, most matched systems at the low to mid range of efficiency levels considered for SDHV applications are of the "coil-only" type, so matched system fan power is based on the "default" value of 0.365 Watts/CFM. Matched systems are also typically rated with a nominal air flow of 400 CFM/ton. SDHV systems differ because while the default value for fan power can be used for the matched system, the rating method requires use of the actual SDHV fan power. Therefore, to estimate the SDHV fan power from the known information, an assumption regarding blower-motor efficiencies is needed[9].

The default value of 0.365 Watts/CFM is based on data obtained many years ago from residential forced air heating systems. At that time, furnace blowers used either shaded pole motors or split capacitor motors less efficient than those made today. The actual value of

[9] Measured fan power from ARI certification test data were not useful because either the static pressures were not available or not at the 1.2 in. H_2O required.

default fan power is a matter of some controversy, with some arguing that the value is too high, and others taking the opposite position. SDHV blowers and blower motors use technologies such as permanent split capacitor (PSC) motors, which generally have higher efficiency. Absent specific efficiency data, the efficiency corresponding to the default fan power of a conventional system is assumed to be 45 percent. Analysis of data shown in Figure 1 for more recent commercially available blowers show a range of blower motor efficiencies from 50 to 80 percent, with a mean of about 65 percent. Consequently, this is the value selected for the SDHV analysis.

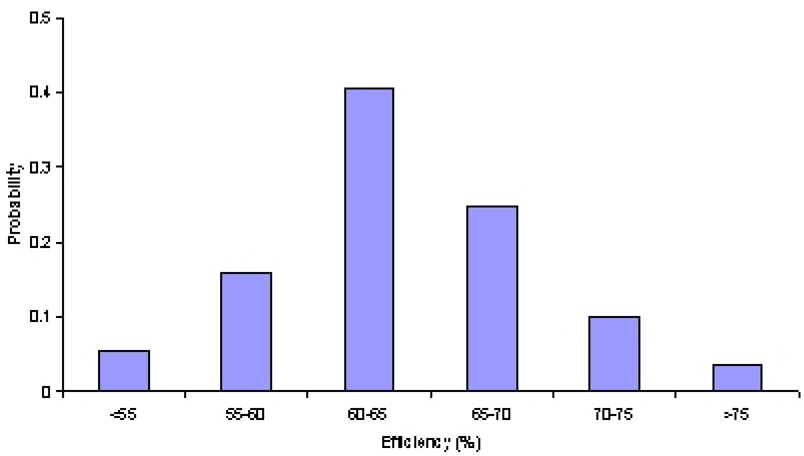

Figure 1: Distribution of Typical Fan Efficiencies for Small-Size Motors (from publicly available data).

Indoor coil scaling factor, $F_c = Q_{c,x}/Q_{c,m}$

In the NIST rating method, described in Appendix A, F_c is the ratio of indoor coil evaporator capacities of the mixed (SDHV) and the matched systems at their respective CFM. Establishing suitable indoor coil scaling factor presents a challenge to predicting mixed system performance ratings, because of the wide variety of matched coil design in current use.

In the technical paper describing the basis of the NIST rating method, system capacity ratio at 95 ºF is proportional to $F_c^{0.37}$ and system capacity ratio at 82 ºF is proportional to $F_c^{0.35}$. System efficiency, or Seasonal Energy Efficiency Ratio (SEER), is directly proportional to the capacity ratio at 82 ºF and inversely proportional to the compressor power ratio at 82 ºF. Since the F_c exponents for capacity ratio and power ratio at 82 ºF are 0.35 and 0.14, respectively, the system efficiency is approximately proportional to $F_c^{0.21}$. Note that, for coil sizes that are typically found in SDHV systems, the correlation predicts system cooling capacity within ±2 percent and SEER within ±1 percent. This is well within testing accuracies.

The use of equation 2.4 is believed to be suitable for the SDHV analysis because it is based on empirical relationships used by designers in predicting heat exchanger performance when changes are made to a coil of known performance, either of coil size or airflow or both. For example, a 10 percent increase in CFM results in a 5 percent increase in coil capacity, and a 10 percent increase in face area with the same CFM also results in a 5 percent increase in coil capacity. Figure 2 demonstrates that the 0.5 exponent in equation 2.4 is reasonable, based on coil manufacturer catalog data.

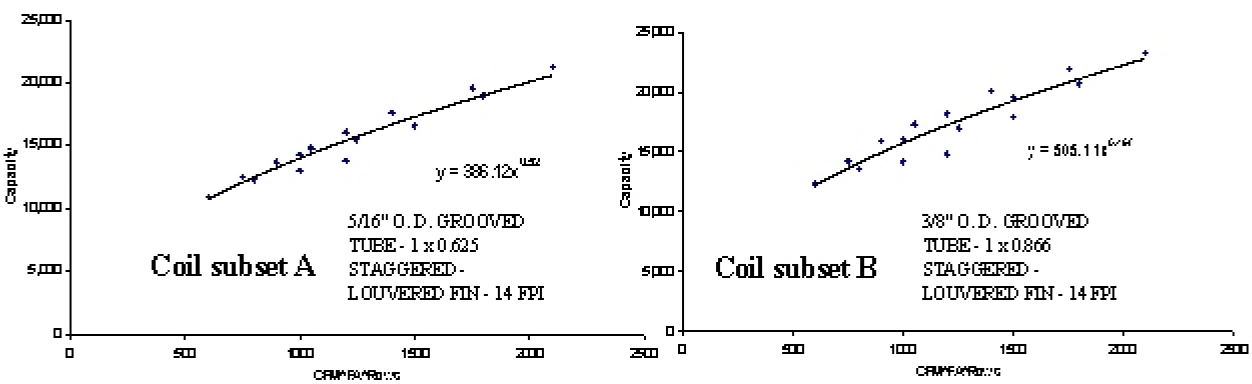

Figure 2: Correlations Between Coil Size and Coil Capacity.

2.3 Description of Baseline SDHV System

The starting point for the analysis of efficiency standards is the definition of a baseline system, which just meets the current minimum efficiency standard of 10 SEER. An examination of certified performance ratings in the ARI directory indicates that a significant number, if not most SDHV systems, just meet the current 10.0 SEER standard. Rather than selecting an actual system, which is considered arbitrary, DOE defined a baseline system assuming the attributes described in Table 1.

Product Group	SDHV Efficiency (SEER / HSPF)	Delivered SDHV Capacity (Btu/hr)	SDHV Airflow Rate (CFM)	SDHV Coil Size (Face Area (sqft × Rows)	SDHV Blower Motor Efficiency	Matched Coil Size (Face Area (sqft × Rows)	Features of Matched Condensing Unit
Air Conditioners	10.0	30,000	800	2 × 4	65%	4 × 3	12.0 SEER - 36,000 Btu/hr

Table 1: Physical Characteristics of Baseline Model.

2.4 Design Options

A SDHV manufacturer has several options available to improve system efficiency. These consist of: a) changes to the design and/or operation of current SDHV blower-coil units; and b) changes to the set of currently available condensing units. In the category of changes to the SDHV unit, the Department includes design changes which a) increase blower motor efficiency; and b) increase coil size. In the category of operation changes, DOE includes adjusting SDHV air flow. In the category of changes to currently available condensing units, the Department includes: a) selecting higher efficiency condensing units; and b) restricting the type of condensing units to those whose matched systems performance ratings are not obtained through use of excessively large indoor coils.

While it is theoretically possible to redesign a condenser unit to optimize the performance for the reduced indoor air flows associated with SDHV systems, the small market share of such systems makes condenser unit redesign an unlikely option.

In the following section DOE discusses in a qualitative sense each of these options and then estimates the costs associated with those options that make sense from a basis of efficiency improvement.

2.4.1 Increased Efficiency of Blower Coil Components

Changes to certain SDHV components can increase the efficiency of a SDHV system. These include improved blowers and improved coils. The efficiency of the blower can be increased in several ways, typically by: 1) substituting a higher efficiency, electronically commutated motor (ECM) for the current permanent split capacitor (PSC) motor; 2) substituting a blower with air foil or backward-inclined blades for the current blower; 3) simply replacing the existing coil in each SDHV blower coil model with a larger coil (either larger face area or more rows).

Design option 1 could potentially be cost effective in modulating systems, because ECMs are substantially more efficient than baseline permanent split capacitor (PSC) motor at partial load. However, because SDHV systems must deliver air to the terminal outlets at a prescribed velocity (to ensure proper mixing with room air) modulation of the blower is not considered to be a reasonable approach and the small efficiency improvement at full load does not justify the much greater cost of the ECM. Option 2 could provide an increase in blower efficiency where space is not an issue, but backward-inclined blowers are considerably larger than conventional forward-inclined blowers and are not a good choice for space-constrained applications, such as SDHV systems. Therefore, DOE did not consider blower-motor design changes as viable options for the purpose of this analysis.

One of the simplest design options which increases SDHV system efficiency is to enlarge the coil's face area. Since current SDHV coil designs are either 4 row or 6 row coils, and the tube and fin designs have near optimal features and configurations, adding face area is the only viable design option. A larger coil has a beneficial effect on both efficiency and capacity. Since current SDHV models have coils which fill available cabinet or module space, the cabinets themselves require enlarging as does the cabinet insulation and external drain pans. Of course, there is a practical limit to how much the indoor cabinet can be increased in size and still stay within field installation limitations.

2.4.2 Reduced Air Flow

For a nominal 3-ton system, the indoor airflow quantity of SDHV systems will range from about 600 to 840 CFM. In principle, when the airflow of the SDHV system is decreased, the F_c ratio also decreases and compressor power decreases. Thus, as airflow is decreased from 840 CFM to 600 CFM, the SEER of the SDHV system increases by about 0.2 SEER, but the system with the lower airflow has a reduced capacity.

Major changes to the range of 600 to 840 CFM for indoor airflow rates at that cooling capacity level should not be considered for SDHV systems. The airflow rates cannot be increased by any substantial amount because the pressure drop through the small ducts would become excessive. The airflow rates cannot be decreased by any substantial amount because the high velocity that is required to properly enter and mix with room air would not be available. Also extending the range in either direction could cause the CFM per rated-ton of the SDHV system to drift outside the mandated limits of 220 to 350 CFM per rated ton.

2.4.3 Selection of Condensing Units

In the category of changes to currently available condensing units, the option available to SDHV manufacturers to increase SDHV efficiency is to either select more efficient condensing units or to restrict the selection of condensing units to those whose matched ratings will not significantly degrade SDHV efficiency.

In the SDHV rating method, if all parameters are kept constant, equation B.2 in Appendix B shows that SDHV efficiency is directly proportional to that of the matched system:

$$SEER_x = SEER_m \times (Q_x/Q_m)_{82} / (P_x/P_m)$$

Therefore, increasing the SEER of the matched system by changing the condensing unit from say a 12 SEER to a 13 SEER, increases the SDHV by 8.33 percent. Since the baseline system has a condensing unit SEER of 12 and an SDHV SEER of 10, a new SDHV system with a 13-SEER condensing unit will have an efficiency of 10.83. However, the capacity ratio and the power ratio are not necessarily constant with SEER, so the coil scaling factor, F_c, must also be considered. Thus, if the improved SEER for the matched system is achieved by use of a larger indoor coil, the improvement in SDHV performance will be lower.

Another option is to be more selective about the matched system indoor coil size. It is possible to select condensing units that achieve matched system SEERs with relatively small indoor coils and thus increase the value of F_c, the coil scaling factor. Similarly, it is possible to avoid condensing units that achieve matched system SEERs with relatively large indoor coils which decreases the value of F_c ratio. However, DOE believes that limiting the selection of condensing units might cause distributors to move away from SDHV products. Therefore, this option negatively impacts SDHV manufacturers in terms of marketing and sale opportunity and is not considered further in the analysis.

2.4.4 Use of Emerging Technologies

DOE did not consider other emerging technologies in this analysis, due to the small size of the small duct, high-velocity system industry. With regards to microchannel heat exchangers, SDHV systems are constrained by the size of the evaporator, which is not currently being considered as a viable application for microchannel technology because of condensate removal concerns.

2.4.5 Max Tech

DOE estimates that the maximum technologically feasible efficiency for SDHV systems is 13.4 SEER. This maximum performance was calculated based on the following two considerations: 1) Condensing units with very high efficiency have features that are unsuitable to SDHV systems. The most efficient condensing units that can be successfully coupled with SDHV blower-coils have a 14-SEER rating; 2) Coil size is limited by the physical constraints that SDHV blower-coils face. Therefore, SDHV coils cannot be larger than the largest coil currently on the market.

2.5 Conclusions on Design Options

Two design options were selected for this analysis: 1) Increased indoor coil size and 2) Higher efficiency condensing unit.

2.6 Estimated Cost Impacts

DOE relied on several sources to estimate the manufacturing costs of the two selected design options: the Residential Air-Conditioner Engineering Analysis; confidential information obtained under non-disclosure agreements between Navigant Consulting, Inc. and the SDHV manufacturers, and estimates based on industry expertise and "rules of thumb".

2.6.1 Increased Coil Size

As coil size increases, more materials and labor are needed and the cost increases. While coil material cost is essentially proportional to the coil size, other components experience cost increases in a similar fashion, including cabinet panels, cabinet insulation and the external drain pan provided in all attic installations. Based on confidential information submitted by the SDHV manufacturers to DOE's contractors, this cost in terms of cost increase per increase in the (Face Area × Rows) value is established. Because of the confidential nature of this data, DOE is not disclosing manufacturer cost increases for larger coil sizes. In the sensitivity analysis, discussed in section 2.8, the impact of incremental coil cost on the cost-efficiency curves are analyzed.

2.6.2 Higher Efficiency Condensing Unit

To estimate the cost of more efficient condensing units, DOE relied on the analysis performed for the residential air conditioner rulemaking. First, DOE evaluated the high volume production costs of 12, 13, and 14 SEER condensing units. In the residential air conditioner analysis, the costs were divided into costs allocated to the outdoor unit, to the indoor unit and to overhead (see Appendix B of the TSD). To determine the costs of the outdoor unit only, DOE added a fraction of the overhead costs, in proportion to the direct costs, to the direct cost of the outdoor unit. For example, if the direct cost of the outdoor unit is 78 percent of the total direct cost, the overhead cost for the outdoor unit is also 78 percent of the total overhead cost. The same procedure was followed for 13 and 14 SEER. The results are presented in the Table 2.

		12	13	14
Outdoor Unit - Direct Costs	Coil Materials	$59.31	$69.63	$94.85
	Coil Labor	$4.76	$4.92	$5.40
	Electrical Materials	$31.99	$35.22	$35.17
	Electrical Labor	$1.07	$1.07	$1.07
	Miscellaneous Materials	$5.66	$5.98	$6.30
	Miscellaneous Labor	$1.73	$1.76	$1.78
	Fan Materials	$4.47	$5.11	$4.86
	Fan Labor	$0.25	$0.25	$0.25
	Cabinet Materials	$18.83	$20.55	$21.88
	Cabinet Labor	$2.45	$2.51	$2.55
	Plumbing Materials	$11.32	$13.55	$14.46
	Plumbing Labor	$3.72	$3.99	$4.21
	Compressor Materials	$152.87	$167.25	$167.78
	Compressor Labor	$0.65	$0.65	$0.65
Overhead Costs	Refrigerant Matl	$5.60	$6.11	$8.08
	Refrigerant Labor	$0.36	$0.36	$0.36
	Indirect Labor	$12.74	$12.85	$15.63
	Indirect Material	$11.28	$10.52	$11.60
	Equipment Depreciation	$5.99	$6.59	$7.47
	Building Depreciation	$11.70	$13.59	$15.47
	Maintenance	$3.07	$3.20	$3.41
	Utilities	$3.82	$4.36	$4.95
	Taxes	$4.21	$4.76	$5.29
	Insurance	$3.74	$4.23	$4.70
	Freight-Out	$45.21	$63.91	$62.31
		$406.80	**$462.91**	**$500.49**

Table 2: Cost Allocation for Outdoor Unit.

However, these costs were obtained assuming that a new standard would be set at the analyzed level and, therefore, do not account for lower production volumes and higher margins that are typical of high-efficiency products. In order to estimate the actual cost of 13- and 14-SEER condensing units, DOE applied additional correction factors, based on our knowledge of the industry. In particular, correction factors of 1.05 and 1.10 were applied to the 13- and 14-SEER condensing units respectively to account for lower production volumes. Also, different gross margins were applied across the efficiency levels, since higher-efficiency units are typically sold at a premium. Gross margins were estimated to be 15, 25 and 35 percent respectively for 12-, 13- and 14-SEER condensing units. Average markups were then applied to estimate the consumer costs. The results of this analysis are summarized in Table 3.

CONDENSING UNIT EFFICIENCY ------------------>	12	13	14
Production cost for high volume production rates	$406.80	$462.91	$500.49
Adjustment for actual production rates	1.00	1.05	1.10
Adjusted production cost	$406.80	$486.06	$550.54
Gross Margin, Percent	15.00	25.00	35.00
Manufacturer's Markup	1.18	1.33	1.54
Selling Price to Distributor	$478.59	$648.07	$846.98
Distributor Markup	1.27	1.27	1.27
Selling Price to Dealer	$607.81	$823.05	$1,075.67
Dealer Markup	1.30	1.30	1.30
Dealer Selling Price	$790.15	$1,069.97	$1,398.37
Sales Tax	1.07	1.07	1.07
Consumer Cost	**$845.5**	**$1,144.9**	**$1,496.3**

Table 3: Condensing Unit Consumer Costs.

In the sensitivity analysis, discussed in section 2.8, the impact of different assumptions on the cost-efficiency curves have also been analyzed.

The cost estimates illustrated in the table above were then validated and confirmed through confidential data submitted by the SDHV manufacturers.

2.7 Distribution Chain Markups

SDHV systems are typically sold in the replacement market. The typical distribution chain for these products is identical to the distribution chain for conventional products in the replacement market. Manufacturers sell products to a distributor (wholesaler), the distributor sells it to the contractor (dealer) and the contractor sells and installs the product to the consumer. Therefore, the total distribution chain markup is the product of four factors: manufacturer markup, distributor markup, contractor markup and sales tax.

SDHV manufacturer markups tend to be higher than markups on conventional systems.

Through its contractors and under non-disclosure agreements, the Department gathered confidential information and estimated a SDHV manufacturer markup. Similarly, the Department estimated other distribution markups, although they are very similar to conventional product markups. Because of the confidential nature of the data, the Department is not disclosing markup values.

2.8 Sensitivity Analysis

The Process Rule requires the Department to consider uncertainty. The initial cost-efficiency relationship does not portray the uncertainty and variability in the assumptions. Uncertainty arises when the precise model parameters cannot be determined. Variability arises when the precise values can be determined, but when they vary among manufacturers, suppliers, or processes. To quantify the uncertainty and variability in the consumer price estimates, DOE used Crystal Ball Pro™ to run Monte Carlo simulation analysis.

Distributions were defined for eleven of the assumptions. All other assumptions were considered point values, either because of low uncertainty, or because any variation would not have significant impact on the results. Uniform distributions represent the assumption that no single value predominates across the range of possible values. Triangular distributions represent that assumption that a particular value does predominate, and that other values are increasingly less likely the farther they are from that value. Table 4 summarizes the ranges considered in the sensitivity analysis.

Variable	*Min*	*Mean*	*Max*	*Type of Distribution*
Matched Coil Size $(FA \times Rows)_m$	10	12	14	Normal
Blower Efficiency (η_{blower})	50%	65%	80%	Triangular
Indoor Airflow Rate of Matched System $(CFM/Ton)_m$	400	412.5	425	Uniform
Incremental Coil Cost ($)	**CONFIDENTIAL**			
Actual Condensing Unit Production Rate Factor at 13 SEER	1.0	1.05	1.1	Triangular
Actual Condensing Unit Production Rate Factor at 14 SEER	1.05	1.1	1.15	Triangular
Condensing Unit Manufacturer Gross Margin at 13 SEER	20%	25%	30%	Triangular
Condensing Unit Manufacturer Gross Margin at 14 SEER	30%	35%	40%	Triangular
SDHV Distribution Markups	**±20% - CONFIDENTIAL**			

Table 4: Variability and/or Uncertainty Ranges for the Assumptions Considered in the Sensitivity Analysis.

To run a Monte Carlo simulation analysis, Crystal Ball™ selects inputs randomly according to the distributions and tracks the effects on price and efficiency. Since two final results were tracked, a new Price/SEER parameter that combines the two values was defined. The result is a bell-shaped probability distribution for the Price/SEER value of each sample. Rather than predicting a single consumer price, the distribution describes the likelihood that the actual consumer price is equal to a predicted value. Thus, the uncertainty in the consumer price estimates can be quantified. Figure 3 illustrates a typical Crystal Ball™ output.

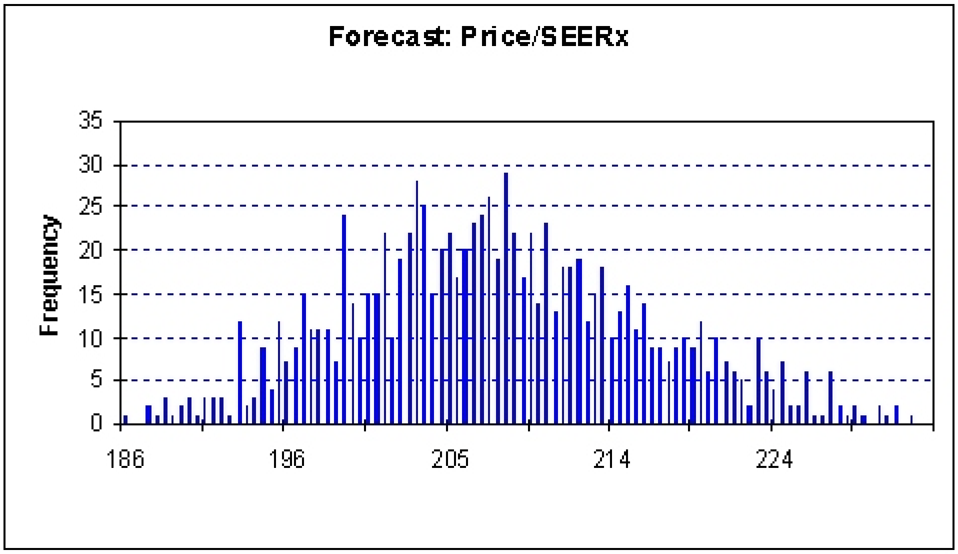

Figure 3: Typical Crystal Ball Output for a SDHV Sample.

On average, Price/SEER values were well within 20 percent of the predicted values with a 95 percent confidence. Therefore, the Department applied a 20 percent uncertainty to the cost-efficiency curve. This uncertainty will be considered in the course of the Life-Cycle-Cost analysis.

2.9 Heat Pumps

Section 4.8.1 of the TSD discusses the relationship between cooling efficiency (SEER) and heating efficiency (HSPF) in air conditioning heat pump systems. Based on heat pump performance data from the Spring 1998 ARI Unitary Directory, DOE proposed the following relationship for heat pumps with cooling efficiencies between 10 SEER and 14 SEER:

$$\text{HSPF} = 6.8 + 0.3 \times (\text{SEER} - 10)$$

In previous proposed and final rules, DOE explained the rational for accepting this relationship. In the May 23, 2002, final rule, DOE adopted this relationship for split system and single packaged heat pumps and for split system and single package through-the-wall air conditioners and heat pumps.

As previously discussed, a SDHV blower coil unit can be operated as a heat pump system by substituting a reverse-cycle heat pump outdoor unit for an air conditioning outdoor unit and by providing the indoor unit with a backup heat source. Currently, one SDHV manufacturer offers a heat pump option. In this case, the coil unit has the same face area but a six-row coil is substituted for a four-row coil to provide the refrigerant management function needed for a heat pump. DOE has no data on the market share of SDHV heat pumps. However, since the common application for SDHV air conditioning systems is in retrofits to existing homes, and since these homes are usually equipped with a hydronic or radiant baseboard heating system, DOE believes that SDHV heat pump market share is small. Consequently, DOE believes that the above SEER vs HSPF relationship adopted for conventional heat pumps and for through-the-wall air conditioners and heat pumps is valid for the purpose of SDHV heat pump standards.

2.10 Air-Distribution Systems and Installation Costs

In our analysis we assume that air-distribution systems and installation costs are constant for all efficiency levels. The Department obtained confidential data from manufacturers and price lists and concluded that the total consumer cost for air-distribution systems is $2720. This cost includes distribution markups and sales tax. In addition, the Department is using the same installation cost that was used for the analysis of conventional systems ($1279).

2.11 A Note on SDHV Cooling Capacity

Previous sections discussed how SDHV manufacturers can improve SDHV system efficiency by increasing the blower-coil size. However, it was not discussed how these changes affect the delivered cooling capacity of the SDHV system. The purpose of this section is to describe how to address the issue of variability in SDHV cooling capacity in our analysis to produce the final cost-efficiency curves.

If a manufacturer simply substituted an existing SDHV coil with a larger coil and kept all other variables constant, the capacity would increase significantly. This is a very basic scenario and is illustrated in the 'Fixed CFM Ratio' (scenario 1) spreadsheet in the published workbook. In this situation, a manufacturer would set the airflow rate at a certain value and let the capacity increase as the coil size increases. DOE used this method to estimate performance values in the baseline case by fixing the SDHV airflow rate to 800 CFM (or, alternatively, fixed the CFM ratio to 2/3).

However, as one increases the size of the coil, there is now extra-capacity that is not needed. In this case, the best approach is to fix the SDHV capacity (at the 30,000 Btu/hr baseline) and let the SDHV airflow rate vary to meet this requirement. SDHV SEER values are typically higher than the 'Fixed CFM Ratio', because the extra-capacity is re-used to increase efficiency. Conversely, costs are lower at a given SEER. This scenario is illustrated in the 'Fixed SDHV Capacity' spreadsheet (scenario 2).

However, by increasing the coil size above a certain value (the threshold varies depending on other assumptions), using the 'Fixed SDHV Capacity' approach, CFM/Ton values fall outside of the definition of SDHV systems (SDHV systems have an airflow rate between 220 and 350 CFM/Ton). Therefore, when hitting the 220 CFM/Ton barrier, the CFM/Ton value was kept constant and all other variables are free of constraints. This scenario is illustrated in the 'Fixed SDHV CFM/Ton' spreadsheet (scenario 3). The capacity will increase as well, but is accounted for in the fact that the manufacturer has no way to charge for this capacity surplus.

Therefore, using a combination of three different scenarios and a combination of the two design options, a cost-efficiency curve was obtained and is shown in section 2.12.

2.12 Summary of Results

The cost-efficiency relationships are summarized in Table 5 and Figure 4. In Table 5, the results are presented in terms of consumer prices at different efficiency levels. A given efficiency level can be reached with different combinations of condensing unit efficiency, which varies from left to right, and SDHV coil size, which varies from top to bottom. For each efficiency level, the least expensive combination was selected. Grey areas represent combinations that are not cost effective. For each combination, the minimum, average and maximum consumer prices are reported.

Condenser Unit SEER ----------->	12				13				14			
		Consumer Price ($)				Consumer Price ($)				Consumer Price ($)		
Coil Size (Face Area (ft^2) x Rows)	SEER	Min	Ave	Max	SEER	Min	Ave	Max	SEER	Min	Ave	Max
8	10.00	0	0	0								
8.5	10.15	21.1	26.3	31.6	11.00	260.6	325.8	390.9				
9	10.29	42.2	52.7	63.2	11.15	281.7	352.1	422.5				
9.5	10.41	63.2	79.1	94.9	11.27	302.8	378.5	454.2	12.14	583.9	729.8	875.8
10	10.51	84.3	105.4	126.5	11.39	323.8	404.8	485.8	12.26	605.0	756.2	907.4
10.5	10.59	105.4	131.8	158.1	11.48	344.9	431.2	517.4	12.36	626.0	782.5	939.1
11	10.65	126.5	158.1	189.7	11.54	366.0	457.5	549.0	12.43	647.1	808.9	970.7
11.5	10.71	147.6	184.5	221.3	11.60	387.1	483.9	580.6	12.50	668.2	835.2	1002.3
12	10.77	168.6	210.8	253.0	11.66	408.2	510.2	612.3	12.56	689.3	861.6	1033.9
12.5	10.82	189.7	237.2	284.6	11.72	429.2	536.6	643.9	12.62	710.4	887.9	1065.5
13	10.87	210.8	263.5	316.2	11.78	450.3	562.9	675.5	12.68	731.4	914.3	1097.2
13.5	10.92	231.9	289.9	347.8	11.83	471.4	589.3	707.1	12.74	752.5	940.6	1128.8
14	10.97	253.0	316.2	379.4	11.88	492.5	615.6	738.7	12.80	773.6	967.0	1160.4
14.5					11.93	513.6	642.0	770.4	12.85	794.7	993.3	1192.0
15					11.98	534.6	668.3	802.0	12.90	815.8	1019.7	1223.6
15.5					12.03	555.7	694.7	833.6	12.95	836.8	1046.0	1255.3
16					12.08	576.8	721.0	865.2	13.00	857.9	1072.4	1286.9
16.5									13.05	879.0	1098.7	1318.5
17									13.10	900.1	1125.1	1350.1
17.5									13.14	921.2	1151.4	1381.7
18									13.19	942.2	1177.8	1413.4
18.5									13.23	963.3	1204.1	1445.0
19									13.27	984.4	1230.5	1476.6
19.5									13.32	1005.5	1256.8	1508.2
20									13.36	1026.6	1283.2	1539.8

Table 5: Consumer Costs as a Function of SDHV Efficiency and Coil Size.

The points of Table 5 can then be connected to build the final consumer cost-efficiency curve, as shown in Figure 4.

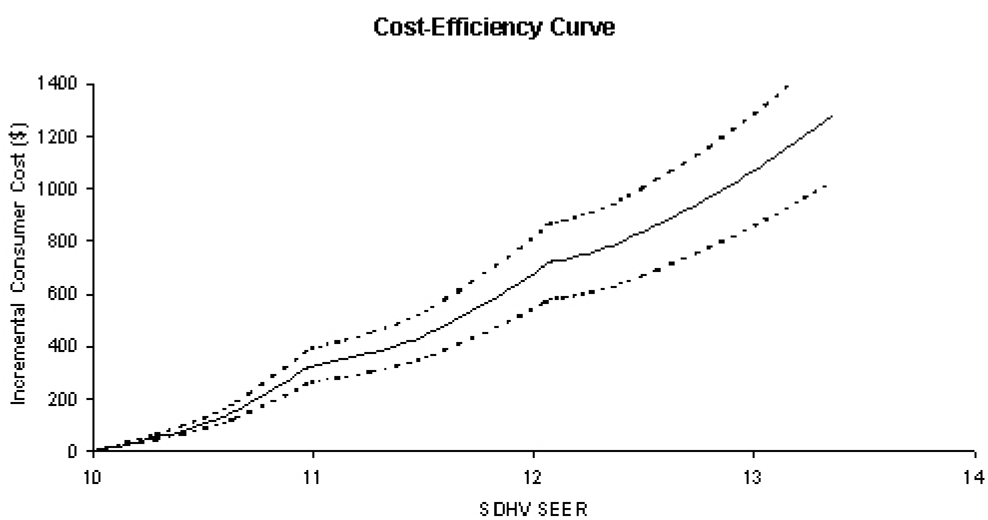

Figure 4: Cost-Efficiency Curve.

APPENDIX A: NIST METHOD FOR RATING MIXED AIR CONDITIONING SYSTEMS

To explain the NIST method for rating mixed air conditioning systems, relevant sections have been excerpted from Chapter 4 of NISTIR 89-4071, retaining all the notation and equation numbering:

"4. PROCEDURE FOR RATING MIXED SYSTEM

4.1 Rating Correlations

Mixed system capacity at DOE Test A condition, Q_x shall be calculated using equation 4.1.

$$Q_x = [Q_m + 3.413 \cdot P_{F,,m}] \bullet F_c^{0.37} \bullet F_e^{\alpha} - 3.413 \; P_{F,,x} \qquad (4.1)$$

Mixed system Seasonal Energy Efficiency Ratio, $SEER_x$, shall be calculated using equations 4.2, 4.3, and 4.4 (derivation of these equations is presented in [2]).

$$SEER_x = SEER_m \; [Q_x/Q_m]_{82} \,/\, [P_x/P_m]_{82}^{-1} \bullet F_{TXV} \qquad (4.2)$$

$$[Q_x/Q_m]_{82} = [1+ (3.25 \bullet P_{F,,m}/Q_m)] \bullet F_c^{0.35} \bullet F_e^{\alpha} - 3.25 \bullet P_{F,,x}/Q_m \qquad (4.3)$$

$$[P_x/P_m]_{82} = 0.8 \bullet F_c^{.14} \bullet F_e^{\beta} + 0.1 \bullet P_{F,,x} \,/\, P_{F,,m} + 0.1 \qquad (4.4)$$

Symbols used in equation 4.1, 4.2, 4.3 and 4.4 are explained below.

<u>Exponents</u>

$\alpha = -0.15$ for $F_e \geq 1$

$\alpha = 0$ for $Fe < 1$

$\beta = 0$ for $F_e \geq 1$

$\beta = -0.2$ for $Fe < 1$

Other Svmbols

F_c = indoor coil scaling factor calculated as explained in section 4.2.1

F_e = expansion device scaling factor calculated as explained in section 4.3.1

F_{rxv} = thermostatic expansion valve factor. (Shall be evaluated as shown in Table 1).

$P_{F.,m}$ = power input to the indoor fan of a matched system as defined in section 4.4.1 (watt).

$P_{F.,x}$ = power input to the indoor fan of the mixed system as defined in section 4.4.2 (watt).

Q_m = capacity of the matched system at Test A as certified by its manufacturer (Btu/h).

Q_x = capacity of a mixed system at Test At as calculated by equation 4.1 (Btu/h).

$[Q_x/Q_m]_{82}$ = ratio of capacities at Test B conditions of the mixed and matched system.

$[P_x/P_m]_{82}$ = ratio of power inputs at Test B conditions of the mixed and matched system.

$SEER_m$ = seasonal energy efficiency ratio of the matched system (Btu/(h- watt) as certified by its manufacturer.

$SEER_x$ = seasonal energy efficiency ratio of the mixed system (Btu/(h-watt) as calculated by equation 4.2.

4.2 Indoor Coil Scaling Factor

4.2.1 Determination of the Indoor Coil Scaling Factor

The indoor coil scaling factor, F_c is defined by the following equation:

$$F_c = Q_{c,x} / Q_{c,m} \tag{4.5}$$

where:

$Q_{c,.x}$ = cooling capacity of a mixed coil at the air mass flow rate specified for the mixed system. The air mass flow rate specified for the mixed system shall satisfy conditions of Appendix M to Subpart B of [6].

$Q_{c,.m}$ = cooling capacity of a matched coil at the indoor-air volumetric flow rate, CFM_m (ft^3/min), at which matched system capacity, Q_m, was measured. If CFM_m information is not available, the value for the indoor air volumetric flow rate shall be calculated as follows:

$$CFM_{m}. = Q_m \bullet 425/ 12000 \ \ (ft^3/min) . \tag{4.6}$$

Capacities of matched and mixed coils shall be obtained using the same verified method (see Section 4.2.3).

Coil capacities shall be obtained at the following conditions:

- inlet air conditions - 80°F dry bulb/67°F wet bulb
- refrigerant saturation temperature at the evaporator outlet - 45°F
- identical refrigerant superheat at the evaporator outlet.

If coil capacities are obtained by means of a catalog or computer simulation, the same catalog or computer simulation shall be used for both coils. Coil material and geometry (e.g., inside tube diameter, tube staggering, fin spacing, fin thickness, fin shape, and number of tube rows) shall be accounted for by the method used. That is, the methodology used must have these parameters as independent variables.

4.2.2 Restrictions

The acceptable range of values for the indoor coil scaling factor, Fc, is from 0.8 to 1.3. This rating procedure shall not be used if the indoor coil scaling factor is smaller than 0.8. If the ratio $Q_{c,x}/Q_{c,m}$ results in a value greater than 1.3, the value of the indoor coil scaling factor, Fc, shall be 1.3.

....

4.4 Power Input to the Indoor Fan

4.4.1 Power Input to the Indoor Fan of the Matched System

Power input to the indoor fan, $P_{f.m}$ shall be measured in accordance with Appendix M to Subpart B of [6], at the indoor-air volumetric flow rate, CFM_m at which capacity of the matched system, Q_m, was measured.

If CFM_m information is not available, the value for the indoor-air volumetric flow rate shall be calculated by equation (4.6). If the indoor fan is not supplied with the system, $P_{f.m}$ shall be evaluated by the equation:

$$P_{f.m} = 0.365 \bullet CFM_m \qquad (4.12)$$

where CFM_m (ft^3/min) is a volumetric flow of air through the matched indoor coil at which system capacity, Q_m, was measured.

4.4.2 Power Input to the Indoor Fan of the Mixed System

Power input to the indoor fan, $P_{f,x}$ shall be measured in accordance with Appendix M to Subpart B of [6], at the indoor volumetric air flow rate, CFM_x, at which capacity of the mixed system, Qx, is evaluated.

If the indoor fan is not supplied with the system, $P_{f,x}$ shall be evaluated by the equation:

$$P_{f,x} = 0.365 \bullet CFM_x \qquad (4.13)$$

where CFM_x (ft^3/min) is the volumetric flow of air through the mixed indoor coil at which the capacity of the mixed system, Q_x, is to be evaluated."

APPENDIX B: PERFORMANCE EQUATIONS

In this section (and in the spreadsheet), nomenclature and terminology used in the NIST method are retained, except the equations are renumbered. As with the NIST method, the subscript notation 'm' indicates the matched system and the subscript 'x' indicates the mixed (SDHV) system. Similar to performance equations 4.1 to 4.4 in NISTIR 89-4071 (Appendix A), the following equations are used:

$Q_x(95)$: Cooling capacity of the mixed (SDHV) system (at 95 °F ambient temperature)

$$Q_x(95) = [Q_m + (3.413 \times P_{F,,m})] \times F_c^{0.37} - (3.413 \times P_{F,,x}), \qquad \text{(B.1)}$$

Note that F_e, the expansion device scaling factor in NIST equation 4.1 is assumed equal to unity.

$SEER_x$: SEER efficiency of the mixed (SDHV) system

$$SEER_x = SEER_m \times (Q_x/Q_m)_{82} / (P_x/P_m)_{82} \qquad \text{(B.2)}$$

Note that F_{TXV}, the thermostatic expansion valve factor in NIST equation 4.2 (Appendix A) is assumed equal to unity.

$(Q_x/Q_m)_{82}$: Ratio of cooling capacities of the mixed (SDHV) system and the matched at 82 °F ambient temperature.

$$(Q_x/Q_m)_{82} = [(1 + 3.25 \times P_{F,,m} /Q_m) \times F_c^{0.35}] - [3.25 \times P_{F,,x} /Q_m] \qquad \text{(B.3)}$$

$(P_x/P_m)_{82}$: Ratio of power inputs of the mixed (SDHV) system and the matched system at 82°F ambient temperature.

$$(P_x/P_m)_{82} = (0.8 \times F_c^{0.14}) + (0.1 \times P_{F,x}/P_{F,m}) + 0.1 \qquad \text{(B.4)}$$

Note that the above expressions for capacity ratio and power ratio are identical to those in the NIST method.

All these equations, along with the assumptions discussed in section 2.2 of this document are included in a calculation spreadsheets that is available on the DOE web site (http://www.eren.doe.gov/buildings/codes_standards/notices/notc0049/index.html).